Thermodynamic Model and Numerical Simulation of Single-Shaft Microturbine Performance

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Abstract: A combined production system based on microturbine holds the promise of increasing energy utilization efficiency and improving environmental quality due to its many attractive merits as a distributed energy source technology. To analyze and evaluate the energy saving potential and economical benefits of microturbine and its combined production system, a simple mathematical model of microturbine is proposed. Part-load characteristics of main components are also considered for analyzing the unit's performance under off-design situations. The proposed model is validated by operational data of a commercially available micro-turbine from a reference. The result shows that the proposed mathematical model can preferably represent the quasi-static operational features of microturbine.

Keywords: Mathematical model; Microturbine; Numerical simulation

1. INTRODUCTION

Due to significantly increasing energy utilization efficiency, decreasing energy use costs, reducing pollutants emission, and improving environmental quality, combined heating and power (CHP) or combined cooling, heating and power (CCHP) has been widely focusing on recent years. There are many research contributions in this field [1-5]. CHP or CCHP systems are economically attractive for many types of buildings, such as commercial buildings, official buildings, hospitals, college campuses, and other buildings which simultaneously have demands for power, heating, cooling and/or other services. Many applications had been reported, such as in supermarket [6], campuses [7], government facilities [8], and medical complex [9].

As the main component of CHP or CCHP system, features of distributed power generation equipments, mainly including efficiency, costs, pollutants emission level, available thermal energy quality, modulability etc., have significant effect on the performance of cogeneration or trigeneration systems. Microturbine is a kind of new emerging gas turbine technology with the features of high rotating speed, low pressure ratio, and modest turbine inlet temperature. Compared with the other DG technologies, microturbine has the advantages of lower initial capital and maintenance costs, modest pollutants emission, higher reliability and relatively lower noise level due to its relatively fewer moving parts.

To accurately analyze and assess the energy saving potential and the economic feasibility of microtubine for distributed power generation and combined production use, optimize operation modes of cogeneration or trigeneration system, a simple but enough accurate performance model of microturbine is desired. Many researchers have engaged on developing mathematic model for gas turbine and many models were developed. A good review about that was given by Jurado [10]. But among these existing models, some of them are detailed first principle models based upon fundamental mass, momentum and energy balances, and thus are very complicated and time-consumed in computation. These models are not suitable for hourly energy consumption analysis of equipment operation though they can be used for design of gas turbine. To simulate the dynamic characteristics of microturbine and design control system for it, some non-linear models were developed by Jurado [10-11] and Zhu [12]. However, these models are mainly interested in electric-mechanical behavior and care few about energy conversion and utilization process.

A piecewise linear model was presented to evaluate the economic and energy saving characteristics of a cogeneration system comprised of microturbine and desiccant air conditioning units [13]. In this model, performance characteristics of microturbine were supposed to be influenced only by its inlet air temperature and other influence factors were neglected. Much useful research work has been done by Zaltash and his cooperators in developing models for building cooling, heating and power system [14~16]. Based on experimental data of a commercially available microturbine, a semi-empirical model was developed by Labinov [15]. In his model, the efficiencies of turbine, compressor and recuperator were regards as constants and thermophysical properties of air and flue gas were assumed to be not change, which is not the case in practice.

The purpose of this study is to build a simple mathematic model for single-shaft microturbine. With this model, it can help designer, operator and manager not only to understand the performance characteristics of microturbine under various operation conditions and thus finally promote the improvement of unit's efficiency, but also to analyze and assess the energy saving and economic feasibility of cogeneration or trigeneration, determine its optimal equipment configuration and operation mode when it is coupled with mathematic models of other combined production equipments.

2.THERMODYNAMIC CYCLE OF SINGLE SHAFT MICROTURBINE

Fig.1 and Fig.2 are respectively the schematic and thermodynamic cycle of microturbine. After cooling the electrical generator, the temperature of ambient air rises up from T_0 to T_1 and the pressure of it drops from P_0 to P_1. Then, it is compressed by compressor to state point 2 with temperature and pressure of T2 and P2 respectively. To improve thermal efficiency and save fuel, compressed air enters into recuperator, where it is preheated from temperature T2 to temperature T3 by hot exhaust gas from turbine. Because of the hydraulic resistance of recuperator, air pressure simultaneously drops from P2 to P3. After recuperator, preheated air goes into combustion chamber, where it combusts with fuel pressurized by fuel compressor and is heated by chemical energy of fuel released during combustion process from temperature T3 to temperature T4. Hot combustion gas with temperature T4 and pressure P4 is fed into turbine. In that component, it expands from state point 4 to point 5 and expansion work is output. Exhaust gas of turbine with temperature T5 and pressure P5 enters recuperator again. After releasing part of its remaining heat into incoming air, its temperature drops from temperature T5 to T6. At the same time, its pressure decreases from P5 to P6 due to hydraulic resistance of recuperator. After that, exhaust gas is finally released into ambient air if there is no thermally activation equipment being sequently installed.

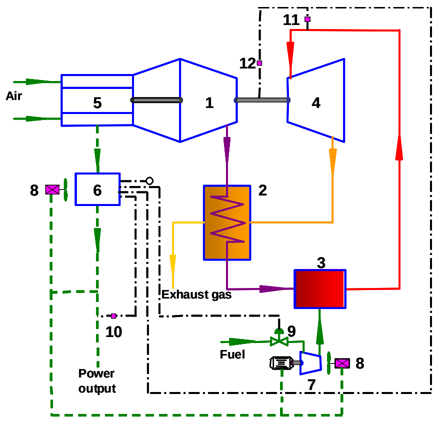

Fig.1 Schematic of single-shaft microturbine

1. Compressor 2. Recuperator 3. Combustion chamber 4. Turbine 5. Generator 6. Controller 7. Fuel compressor 8. Cooling fan 9. Regulating valve 10. Power sensor 11. Thermostat 12. Rotation speed sensor

In Fig.2, there are two empty dots, point $2'$ and point $5'$, which are not the state points of thermodynamic cycle but two supplement points. Point $2'$ and point 2 have the same pressure but different temperature. Point 2 is the final state of compressed air after undergoing an actual polytropic compression process with a pressure rising from P1 to P2 while point $2'$ is the final state of air after undergoing an ideal isentropic compression process with the same initial and final pressures. Similarly, point 5 is the air state point after undergoing an actual polytropic expansion process while point $5'$ is that after undergoing an ideal isentropic expansion process with the same initial and final pressures.

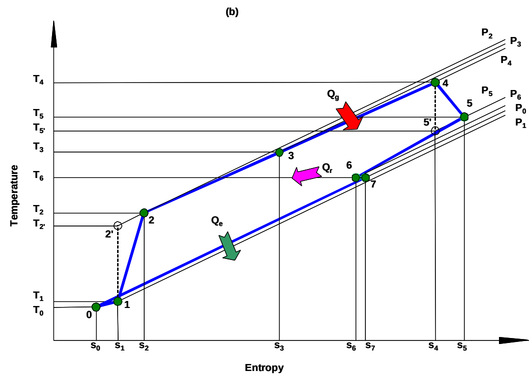

Fig.2 Thermodynamic cycle of microturbine

3.MATHEMATICAL MODEL OF MICRO-TURBINE

Each thermodynamic process of microturbine is analyzed based on assumptions as followed.

(1) Air and combustion products are treated as perfect gases.

(2) System operates under normal operating conditions. Start-up, shut-down, and other fast dynamic processes are not included.

(3) Thermodynamic and flow processes progress along the steady state performance curves, that is, quasi-static processes are considered.

(4) Energy storage and transport delay in various components of microturbine are all neglected, thus, steady state equations are applied.

(5) Flow in unit is regards as a one-dimensional flow process. Flowing kinetic energies of air and combustion gas is treated as negligible.

3.1 Incoming Duct (process 0-1)

The thermodynamic process of incoming duct can be represented by Eq.(1) and Eq.(2).

$$Q_{0-1} = G_a(h_1 - h_0) \tag{1}$$

$$P_1 = P_0 - \xi_{0-1}\frac{G_a^{\ 2}}{2\bar{\rho}_{a,0-1}} \tag{2}$$

where, ξ is the flow resistance factor of duct; $\bar{\rho}_a$ is average density of air in duct; P is air pressure ;

Q_{0-1} is heat added in the process of 0 to 1; G_a is air mass flow rate; h is specific enthalpy of air. Subscript 0 or 1 denotes air state shown as Fig.2 and subscript 0-1 denotes the thermodynamic process from state 0 to state 1. Nomenclatures in later equations have similar mean.

3.2 Compressor (process 1-2)

Pressure ratio of compressor is defined as

$$\pi_C = \frac{P_{r2}}{P_{r1}} = \frac{P_2}{P_1} \tag{3}$$

where, P_r is relative pressure. Isentropic efficiency of compressor is defined as

$$\eta_{C,S} = \frac{h_{2'} - h_1}{h_2 - h_1} \tag{4}$$

where, $h_{2'}$ is the specific enthalpy of air after undergoing isentropic compression process, which can be calculated using relative pressure. If $\eta_{C,M}$ denotes the mechanical efficiency of compressor, the practical power to drive compressor can be determined by following equation.

$$W_C = \frac{G_a(h_2 - h_1)}{\eta_{C,M}} \tag{5}$$

3.3 Recuperator (process 2—3 and 5—6)

In microturbine, recuperator is used to reduce fuel consumption and improve thermal efficiency though preheating incoming air by hot exhaust from turbine. Recuperation efficiency of recuperator is defined as

$$\sigma = \frac{T_3 - T_2}{T_5 - T_2} \tag{6}$$

where, T is air temperature. Heat recovered by recuperator can be calculated by following equation.

$$Q_R = G_a(h_3 - h_2) = G_g(h_5 - h_6)\eta_R \tag{7}$$

where, η_R is adiabatic efficiency which indicates magnitude of heat loss of recuperator into ambient air; G_g is the mass flow rate of combustion gas. Outlet pressure of incoming air and inlet pressure of exhaust gas are determined by Eq.(8) and (9) respectively.

$$P_3 = P_2 - \xi_{2-3}\frac{G_a^{\ 2}}{2\overline{\rho}_{a,2-3}} \tag{8}$$

$$P_5 = P_6 + \xi_{5-6}\frac{G_g^{\ 2}}{2\overline{\rho}_{g,5-6}} \tag{9}$$

3.4 Fuel Compressor (process 8—9)

When the pressure of fuel distribution net is not high enough, microturbine need equip fuel compressor to pressurize fuel and supply fuel into combustion chamber. Thermodynamic process of fuel compressor is similar to that of air compressor. Compression ratio of fuel compressor is defined as

$$\pi_{FC} = \frac{P_{r9}}{P_{r8}} \tag{10}$$

Isentropic efficiency of fuel compressor is defined as

$$\eta_{FC,S} = \frac{h_{9'} - h_8}{h_9 - h_8} \tag{11}$$

Power consumed by fuel compressor is calculated by followed equation.

$$W_{FC} = \frac{G_f(h_9 - h_8)}{\eta_{FC,M}} \tag{12}$$

3.5 Combustion Chamber (process 3—4)

During the process of fuel combusting with air, the chemical energy of fuel is released though oxidization action. Under adiabatic condition, energy released during combustion process is completely absorbed by combustion products consequently increasing their temperature. The maximum temperature the products can achieve is called as theory combustion temperature. The theory combustion temperature can be computed according to the law of energy conservation. That is

$$H_{\mathrm{Re}a} = H_{\mathrm{Pr}o} \tag{13}$$

where, $H_{\mathrm{Re}a}$ and $H_{\mathrm{Pr}o}$ denote total enthalpy values of reactants and products respectively. In this paper, natural gas is used as fuel of microturbine. Thermophysical properties of methane are used as a replacement of that of natural gas. $H_{\mathrm{Re}a}$ and $H_{\mathrm{Pr}o}$ can respectively be computed by using Eq.(14) and (15).

$$\begin{aligned} H_{\mathrm{Re}a} = &\left[(h_{298}^0)_{CH_4} + M_{CH_4}\int_{298}^{T_9} C_{p\_CH_4}dT\right] + 2\alpha M_{O_2}\left[(h_{298}^0)_{O_2} + \int_{298}^{T_3} C_{p\_O_2}dT\right] \\ &+ 7.52\alpha M_{N_2}\left[(h_{298}^0)_{N_2} + \int_{298}^{T_3} C_{p\_N_2}dT\right] \end{aligned} \tag{14}$$

$$\begin{aligned} H_{\mathrm{Pr}o} = &\left[(h_{298}^0)_{CO_2} + M_{CO_2}\int_{298}^{T_{4,ideal}} C_{p\_CO_2}dT\right] + 2M_{H_2O}\left[(h_{298}^0)_{H_2O} + \int_{298}^{T_{4,ideal}} C_{p\_H_2O}dT\right] \\ &+ 2(\alpha - 1)M_{O_2}\left[(h_{298}^0)_{O_2} + \int_{298}^{T_{4,ideal}} C_{p\_O_2}dT\right] + 7.52\alpha M_{N_2}\left[(h_{298}^0)_{N_2} + \int_{298}^{T_{4,ideal}} C_{p\_N_2}dT\right] \end{aligned} \tag{15}$$

In Eq.(14) and Eq.(15), h_{298}^{0} denotes the standard formation enthalpy; M is molecular mass; C_P is constant pressure specific heat; α is air excess factor. Coupling Eq.(13) to (15), theory combustion temperature, $T_{4,ideal}$, can be computed via iteration. Because $T_{4,ideal}$ is computed under the assumption of complete combustion, to consider the effect of incomplete combustion, a combustion efficiency, which indicates the completeness of combustion, is defined as

$$\eta_{CC} = \frac{T_4}{T_{4,ideal}} \tag{16}$$

Besides energy conservation, mass is also conservation during combustion. That is

$$G_g = G_a + G_f \tag{17}$$

Air excess factor of combustion chamber is determined by followed equation.

$$\alpha = \frac{1}{9.52}\frac{G_a}{G_f}\frac{M_f}{M_a} \tag{18}$$

Outlet pressure of combustion chamber is calculated by Eq.(19).

$$P_4 = P_3 - \xi_{3-4}\frac{G_g^{\ 2}}{2\overline{\rho}_{g,3-4}} \tag{19}$$

3.6 Turbine (process 4—5)

Turbine is the component where hot combustion gas expands to drive turbine rotating and to output mechanical work. Ability of microturbine outputting work is affected by expansion ratio and isentropic efficiency of turbine, which are defined as Eq.(20) and Eq.(21) respectively.

$$\pi_T = \frac{P_{r4}}{P_{r5}} = \frac{P_4}{P_5} \tag{20}$$

$$\eta_{T,S} = \frac{h_4 - h_5}{h_4 - h_{5'}} \tag{21}$$

where, $h_{5'}$ is the specific enthalpy of flue gas undergoing isentropic expansion process, which can be worked out with relative pressure. If $\eta_{T,M}$ is used to express mechanical efficiency of turbine, practical output power of turbine can be calculated by using Eq.(22).

$$W_T = \frac{G_g(h_4 - h_5)}{\eta_{T,M}} \tag{22}$$

3.7 Exhaust Gas Duct (process 6—7)

Due to hydraulic resistance of exhaust gas duct, backpressure of turbine will increase and net power output of microturbine will decrease. Flow process of flue gas in exhaust gas duct can be represented by following equation.

$$P_7 = P_6 - \xi_{6-7}\frac{G_g^{\ 2}}{2\overline{\rho}_{g,6-7}} \tag{23}$$

3.8 Generator

A large fraction of shaft work inputted into generator is converted into electricity by generator and the remaining part is dissipated into surrounding by waste heat. If η_G represents the efficiency of generator, power output of it is

$$W_G = \eta_G(W_T - W_C) \tag{24}$$

Waste heat produced by generator which is finally removed by incoming air, can be calculated by Eq.(25).

$$Q_{0-1} = (1-\eta_G)(W_T - W_C) \tag{25}$$

3.9 Controller

Controller converts the variable-frequency power from the generator into constant-frequency power the users demanded. Due to the existence of electrical resistance, a part of electricity is dissipated in the process of power conversion. Heat generated in the process of conversion is removed by cooling fans. If η_{CN} represents the conversion efficiency of controller, the output of controller is

$$W_{CN} = \eta_{CN} W_G \tag{26}$$

3.10 Work and Heat Balance of Microturbine

The net power output of microturbine can be determined by the following equation.

$$W_{net} = W_{CN} - W_{FC} - W_{CF} \tag{27}$$

where, W_{FC} and W_{CF} are power consumptions of fuel compressor and cooling fans respectively. The overall thermal efficiency and the specific fuel consumption of microturbine are respectively defined as

$$\eta_{overall} = \frac{W_{net}}{G_f Q_f} \tag{28}$$

$$SFC = \frac{G_f}{W_{net}} \tag{29}$$

where, Q_f is higher heating value (HHV) of fuel, which is 50050kJ/kg for natural gas in this research.

4. PART-LOAD PERFORMANCE OF MICRO-TURBINE

Due to change of load and/or ambient condition, microturbine often runs under off-design situation. Therefore, it is significant to study part-load behavior of microturbine. However, the understanding of part-load behavior, especially some quantitative rules, is far from enough due to its complexity. In addition, related experimental data is rather scarce for the commercial sake. All these reasons make it difficult in quantitative analysis of part-load performance of microtuebine. After repeated deliberation, the general performance characteristics formulas of compressor and turbine under part-load condition were proposed and used in investigating the performance of a gas turbine-based cogeneration sets by Zhang, Wang and Cai [17~18]. In this paper, their proposed model is used to study the performance of microturbine under part-load conditions. Heat transfer efficiency of recuperator is affected by flow rates of air and flue gas passing it. Therefore, recuperation efficiency varies with microturbine's output load. In this research, part-load performance of recuperator is represented by Eq.(30).

$$\sigma = \frac{\sigma_0}{\sigma_0 + (1-\sigma_0)(\dfrac{G}{G_0})^{0.2}} \qquad (30)$$

5. VALIDATION OF MATHEMATICAL MODEL

To verify the accuracy of the mathematical model proposed above, operation data of a commercial available microturbine from reference [15] is used to validate the model.The commercial available micro- turbine is a three-phase 480-VAC/30-kW unit with a maximum net power output of 28 kW. The maximum rotation speed of microturbine is 96000 rpm and maximum inlet temperature of turbine is 1116 K. The overall thermal efficiency of unit based on higher heating value (HHV) of fuel is approximately 23.6% under standard operation condition. Information provided by the microturbine manufacturer and some other simulation conditions are given in Table1.

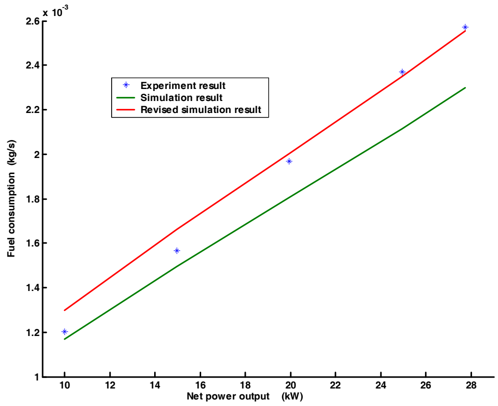

Fig.3 Comparison among experimental data, simulation results and revised simulation results

With parameters in Table 1, the commercial microturbine is simulated by using the proposed mathematical model in this paper. Fig.3 shows the comparison of fuel consumptions between experi-mental measurements and simulation results. From Fig.3, it can be found that the simulation results and experimental results have identical variation trend. With increase of power output, fuel consumption of microturbine increases in a fairly linear fashion. However, it can also be noted from Fig.3 that the simulation results are all lower than the practical fuel consumptions. Reason resulted in this phenomenon is that the fuel used in simulation is slightly different from that used in experiment. The fuel used in simulation is pure methane while that in experiment is natural gas, only about 90% of which is methane. Therefore, the practical fuel consumption is larger than the simulated result. A remedy is made on simulation results through dividing it by 0.9. Revised simulation results are plotted on Fig.3, too. Fig.3 shows that revised simulation results are well accordant with experimental results.

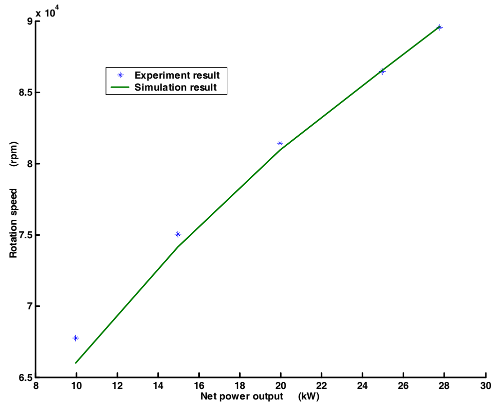

Fig.4 Comparison between measured and simulated rotation speeds

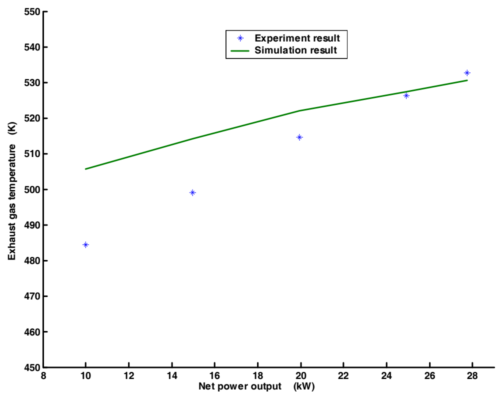

Fig.5 Comparison between measured and simulated exhaust gas temperatures

Fig.4 is the comparison between the practical and the simulated rotation speeds. Fig.4 shows that simulated rotation speed is very consistent with practical rotation speed. Comparison of measured and simulated exhaust gas temperatures are shown in Fig.5. Fig.5 illustrates that simulated results accords with experimental results under design or near-design load condition. However, there exists a relatively larger error between measured results and simulated results under lower load condition than that under design load condition. Measured data is smaller than simulated result. Reason causes this error is maybe the not enough accurate part-load performance model of recuperator. This can also be verified from Fig.3 and Fig.4, which both illustrate a larger error under lower load than under higher load.Result of model validation shows that the proposed mathematical model can basically simulate the operation characteristics of microturbine so that it can be used to analyze operation performance of microturbine though some error exists between experimental and simulated results.

6. CONCLUSION

To help analyzing and evaluating the energy and costs saving potentials in whole life cycle of microturbine, a simple mathematical model is proposed based on analyzing the thermodynamic and flow processes of each component. The proposed model is validated by the operation data of a commercially available microturbine. The result of model validation shows that the proposed mathematical model can preferably represent the quasi-static operational features of microturbine. It is helpful in understanding the performance characteristics of microtubine and can be used for hour-by-hour analyzing and assessing the energy saving potential and economic benefits of it and combined production system based on it. In the part II of this series of papers, the model will be used for performance simulation of microturbine.

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Table 1 Designed parameter of microturbine and simulation condition

Parameter	Description	Value	Parameter	Description	Value
T_{40}	Designed inlet temperature of turbine （K）	1116	π_{C0}	Designed compressor pressure ratio	3.4
G_{C0}	Designed compressor air flow rate（kg/s）	0.31	W_{net}	Net power output of microturbine（kW）	28 ± 1
n_0	Designed engine speed （r/min）	96000	η_{total}	Thermal efficiency of unit （based on HHV）	23.6 ± 1.8
η_{CN}	Efficiency of controller	94%	W_{CF}	Power of cooling fan （kW）	0.122
$\eta_{C,S0}$	Isentropic efficiency of compressor	79%	$\eta_{T,S0}$	Isentropic efficiency of turbine	80%
σ_0	Recuperation efficiency of recuperator	80%	η_{CC}	Combustion efficiency	98%
$\eta_{FC,S}$	Efficiency of fuel compressor	80%	η_G	Efficiency of generator	96%
η_R	Adiabatic efficiency of recuperator	98%	$\eta_{C,M}$	Mechanical efficiency of compressor	100%
$\eta_{T,M}$	Mechanical efficiency of turbine	100%	$\eta_{FC,M}$	Mechanical efficiency of fuel compressor	100%
T_{10}	Designed ambient temperature（K）	288	P_{10}	Designed ambient pressure（Pa）	101325